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Title:

Evaporation process control used in refrigeration

5 **Technical field:**

Evaporation of refrigerant in cooling and freezing
plants, refrigeration engineering, refrigeration
machine for cooling and heating operation,
refrigeration plants, refrigeration sets, heat pumps,
10 air-conditioning systems and others.

Prior art:

Evaporator control with drive dry expansion on the
basis of the minimum stable signal (MMS) (Fig. 1, 2
15 and 3).

For optimum operation of an evaporator used in
refrigeration, the evaporator is supplied with
sufficient wet steam for a control valve (expansion
20 valve) (3) to be controlled to a minimum stable signal,
normally on the basis of the evaporator outlet pressure
(12) and the associated evaporator outlet temperature
(13) (drawing Fig. 1, 2 and 3). The difference between
the evaporator pressure, converted by calculation into
25 the associated evaporation temperature, and the actual
evaporation temperature measured is used as measured
variable for the control valve. In this context, the
aim is stable control characteristics with as low a
temperature difference as possible. As low a
30 temperature difference as possible leads to a higher
evaporator power. If the difference is too low or the
signal is not stable, liquid shocks or power reduction
occur at the compressor (1). If the difference is too
great, the evaporator power (4) is reduced.

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Automatic valves, capillary tubes or other equipment
are also dimensioned and used on the basis of the same

principle (superheated refrigerant vapor at the end of
the evaporation process).

Nowadays, in some cases internal heat exchangers (IHEs)
5   (5) (Fig. 4, 5, 6) are connected downstream of the
evaporator. However, these internal heat exchangers are
designed as "thermally short" equipment and are not
incorporated in the evaporator control on the basis of
the entry vapor content. The refrigerant liquid is not
10 strongly cooled and the suction vapors are not strongly
superheated. The superheating of the suction vapor is
restricted to approx. 5-10K. Injection valves which are
customary nowadays are also not designed for maximum
superheating, and the superheating which can be set is
15 at most approx. 20-25K.

Detailed description of the invention:
It is an object of the invention to achieve the
following in cooling/freezing plants, refrigeration
20 machines for cooling and heating operation,
refrigeration plants, refrigeration sets, heat pumps,
air-conditioning systems and all other systems using
refrigerant for evaporation:

25 To keep the suction vapor superheating in the
evaporator (4) at a low level or to leave the
evaporator (4) with wet steam, and in this case keeping
the suction vapor superheating upstream of the
compressor (1) as high as possible (as far as the use
30 limits of the compressor, the oil or the refrigerant
and/or the various temperature ratios permit).

For this purpose, the refrigeration plant, which
primarily comprises compressor (1), condenser (2),
35 injection valve (3) and evaporator (4), is provided
with an additional internal heat exchanger (5),
referred to below as IHE (Fig. 7, 8, 9, 10, 11).

This IHE (5) is installed between evaporator (4) and compressor (1), on one side, and between condenser (2) and injection valve (3) on the other side (drawing Fig. 8, 9, 10).

On one side, liquid refrigerant flows through the IHE (5) (liquid side), and on the other side superheated refrigerant in vapor form or wet steam flows through the IHE (5).

If pure media (liquid refrigerant and superheated suction vapor) flow through the IHE, it is possible to speak of heat exchange (Fig. 4, 5, 6). If the IHE is operated with a liquid refrigerant and wet steam with subsequent suction vapor superheating, it is possible to speak of a second evaporation stage with integrated liquid supercooling and suction vapor superheating (Fig. 7, 8, 9, 10). The following text always encompasses both possible options.

The actual evaporation (first stage) (4) takes place partly or completely in the evaporator (4). To allow optimum operation of this evaporator (4), liquid refrigerant is admitted at the evaporator outlet.

Since liquid refrigerant is admitted at the evaporator outlet, for control of the evaporator (4) there is an absence of a measurement variable for determining the superheating, and the expansion valve (3) can no longer control the filling of the evaporator (4) with refrigerant.

The control for which a patent is hereby applied for the first time, as a novel feature, makes use of the measurement variables comprising the liquid temperature of the refrigerant upstream of the injection valve (3) and the evaporator pressure (Fig. 7, 8, 9, 10, 11, points 9, 10, 11, 12).

It is in this context irrelevant what types or designs of evaporators and what refrigerants and application areas are involved.

The evaporator pressure is preferably taken at the inlet of the evaporator (12) (start of evaporation) (Fig. 7, 8, 9, 10, 11, point 12). In special cases, the exit pressure or any desired value derived from the two pressure measured values (refrigerant glide) can also be used as measured value (Fig. 7, 23).

This control controls the start of the evaporation process (Fig. 7, points 11, 12) rather than, as has hitherto been the case, the end of evaporation (Fig. 3, points 12 and 13).

It is in this context irrelevant whether control is set to precisely the left-hand limit curve between refrigerant liquid and refrigerant wet steam in the lg p, h diagram of the refrigerant or to a value (to the left) or to the right of this limit curve.

With "optimized" evaporator designs, the evaporation process is started as close as possible to the left-hand limit curve of the lg p, h diagram. In the case of non-optimized evaporators, it may be advantageous for a certain proportion of gas to be admitted at the start of the evaporation process. In this case, the evaporation process is started to the right of this limit curve after the optimum for the respective evaporator.

The start of the evaporation process can be defined by the liquid temperature upstream of the injection valve (11, 9) and the evaporation pressure (12, 10) (Fig. 7, 8, 9, 10, 11, points 11, 12, 9, 10).

The control variable can be defined, and the superheating controlled, from the evaporation pressure and a fixed (temperature) difference (adjustable) or from a stored curve calculation, depending on the refrigerant.

The injection valve (3) lowers the temperature of the refrigerant liquid (11) upstream of the injection valve (3) by opening the valve (3), and increases the refrigerant liquid temperature by closing the valve (3), thereby seeking to keep the desired value at a corresponding evaporation pressure (12).

The degree of flooding or superheating (19, 13) of the evaporator(s) (4) therefore determine the supercooling temperature of the liquid refrigerant (11) at a corresponding evaporation pressure (12) and the suction vapor temperature (13) at the compressor inlet (14).

When limit values are reached, such as for example the maximum permissible temperature for the compressor (13, 14, 15, 16), a further temperature-measuring sensor (optional) takes over and overcontrols the control of the refrigerant liquid entry temperature into the injection valve (11) on the basis of evaporator pressure (12) (Fig. 7, 9, 11, points 11, 12 and 13 (14, 15, 16)).

It is in this context irrelevant whether the suction vapor temperature at the exit of the IHE (5) (13), the suction vapor temperature at the compressor inlet (1) (14), the hot-gas temperature (compressor exit) (15), the oil temperature of the compressor (1) (16) or another suitable temperature is used as measurement variable for this safety and optimization function (Fig. 8, 9, 10, 11, points 13, 14, 15, 16).

In any event, an optimum-maximum supercooling (11) of the refrigerant liquid and an optimum-maximum suction

vapor superheating (14) as a function of the corresponding compressor is always the aim, as a function of the evaporator type (Fig. 7, 9, 10, 11, points 11, 14).

It is in this context irrelevant whether the refrigeration system comprises one or a plurality of evaporators (4), one or a plurality of IHEs (5), one or a plurality of compressors (1) or one or a plurality of injection valves (3), and whether or not they are combined to form groups. It is in this context also irrelevant whether or not one or more evaporators (4) are combined into groups with only one or more IHEs (5) (Fig. 10-18, points 9, 10, 13, 14, 15, 16). Any combinations of injection valves (3), evaporators (4), IHEs (5) and compressors (1) is therefore possible.

It is irrelevant whether the injection valves (3) are of mechanical, thermal, electronic or other design and whether they control cyclically, continuously or in some other way. What is crucial is the process and control circuit, with the dependent relationships which have been listed between start of evaporation (11, 12), end of evaporation (13, 19) as a function of the refrigerant liquid entry temperature (21) to the IHE (5), the suction vapor exit temperature (13) from the IHE (5), the state of the refrigerant (wet steam (19) or superheated suction vapor (13)) on leaving the evaporator (19) and entering (20) the IHE (5), which in one case is operated as a second evaporator stage with subsequent high suction vapor superheating (13) and in another case, in the same plant, is operated as a pure heat exchanger for superheating the suction vapor (13). In this context, it also irrelevant whether an external supercooler stage (25) connected upstream of the IHE (5) is connected to or disconnected from the process.

The advantage of this evaporator control consists in

the fact that in this way the evaporator (4) is
optimally flooded and utilized (drawing Fig. 7, 9, 10,
11, points 17, 19), that the pressure drop on the
refrigerant side across the evaporator (4) is reduced,
5   that as a result the evaporation temperature (23) is
increased, that as a result smaller evaporators (4) can
be used, that as a result the mass flow of refrigerant
for a required refrigeration power is reduced, that as
a result the compressors (1) are smaller (refrigeration
10 production), that as a result less energy is required
for the generation of refrigeration, that as a result
efficiencies and the lubrication and therefore the
service life of the compressors (1) are increased.

15 The control is set in such a way that the maximum power
is always in favor of the evaporator (4) (Fig. 7, 8, 9
points 17) and not the IHE (5) (18) (maximum possible
enthalpy distance at point 17).

20 **Novelty:**

A novel feature of our invention is that an evaporation
system with dry expansion is used as flooded evaporator
(4), in which the refrigerant leaves the evaporator (4)
25 in the first stage with liquid fractions (17, 19).

A novel feature of our invention is that the
refrigerant enters a second evaporation stage (5, 18,
20) (dry evaporator) as a liquid/gas mixture with a
30 high gas content, and residual evaporation with
subsequent high superheating of the refrigerant (13)
and simultaneous supercooling of the liquid refrigerant
on the second side of the IHE (5) takes place in this
second evaporation stage (11).

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A novel feature of our invention is that control is based
on the start of evaporation (12) of the evaporation
process and not on the end of evaporation (13).

A novel feature of our invention is that this control is run on the evaporator (1) with different suction vapor superheating levels (13) depending on the liquid
5   entry temperature (21) to the IHE (5).

A novel feature of our invention is that the suction vapor superheating (13) is selected to be as high as possible.
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A novel feature of our invention is that the expansion valve (3) used, which is installed outside or inside the evaporator, controls the refrigerant liquid temperature (11) before it enters the injection valve
15   (3).

A novel feature of our invention is that the expansion valve (3) used, which is installed outside or inside the evaporator (4), limits the suction vapor
20   temperature at the entry to the refrigerant compressor (14) and at the same time controls the power of the internal supercooling (18) as a function of the evaporator power (17) available from the first stage (4).
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List of drawings:

   • Fig. 1 shows the refrigerant circuit in the "prior art" lg p, h diagram
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   • Fig. 2 shows the "prior art" refrigerant circuit

   • Fig. 3 shows the refrigerant circuit in the
35   lg p, h diagram with integrated equipment

   • Fig. 4 shows the refrigerant circuit in the lg p, h diagram with IHE of the "prior art"

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- Fig. 5 shows the refrigerant circuit with IHE of the "prior art"

- Fig. 6 shows the refrigerant circuit with IHE of the "prior art" in the lg p, h diagram with integrated equipment

- Fig. 7 shows the refrigerant circuit in the lg p, h diagram with two-stage evaporator of the "patent"

- Fig. 8 shows the refrigerant circuit with two-stage evaporator of the "patent"

- Fig. 9 shows the refrigerant circuit in the lg p, h diagram with two-stage evaporator of the "patent" with integrated equipment

- Fig. 10 shows the refrigerant circuit in the lg p, h diagram with two-stage evaporator of the "patent" with integrated equipment and two-stage supercooling (and deheater)

- Fig. 11 shows the refrigerant circuit with evaporator and measured value combinations (example)

- Fig. 12 shows the legend for the points from the drawings

Embodiment of the invention:

A refrigeration system substantially comprising one or more:

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Liquefiers (2), evaporators (4), IHEs (5), refrigerant compressors (1), injection valves (3), refrigerants, refrigeration auxiliary substances and oil.

10   A refrigeration system, depending on its application, optionally also has one or more of the abovementioned components and, in addition deheaters (24), one or more waste heat utilization exchangers, further supercoolers (25), viewing windows (7), driers (6), filters, valves

15   (8), safety equipment, shut-off equipment, accumulators, oil pumps, distribution systems, electrical and control parts, refrigeration auxiliary substances, etc.

20   When fitting the injection valve (3) upstream of the evaporator (4), the measured value for limiting suction vapor is taken off at the suction line leading to the refrigerant compressor (1). The measured values for the refrigerant liquid temperature (11) and the evaporator

25   entry pressure (12) are used to control the evaporation (17, 19).
Alternatively, the measured values for the high pressure (22) upstream of the injection valve (3) and for the suction vapor pressure (12) downstream of the

30   injection valve (3), as well as the hot-gas temperature (15) downstream of the compressor (1) or the oil temperature (16) of the latter, are likewise available for controlling the evaporator (4) with downstream IHE (5).

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Patent Claims

1.   A method for controlling evaporators (4) in
refrigeration plants, characterized in that control is
carried out after the evaporation process (12) has
begun, with the evaporation pressure (12) at the inlet
of the evaporator (4) (12) normally being used as one
control variable and the refrigerant supercooling
temperature (11) upstream of the injection valve (3)
being used as second control variable, so that in this
way the start of evaporation (12) is defined and
controlled.

2.   The method for controlling evaporators (4) in
refrigeration plants as claimed in claim 1,
characterized in that an internal heat exchanger IHE
(5) is connected downstream of the evaporator (4).

3.   The method for controlling evaporators (3) in
refrigeration plants as claimed in claims 1-2,
characterized in that a further measured value, the
suction vapor temperature (13/14) at the compressor
inlet (1), optimizes this control and ensures
protection for the compressor (1) (optional).

4.   The method for controlling evaporators (4) in
refrigeration plants as claimed in claims 1-3,
characterized in that further measured values, such as
the hot-gas temperature (15) at the exit of the
compressor (1), the compressor oil temperature (16),
the suction pressure at the compressor (23) and/or the
high pressure (22) upstream of the injection valve (3)
or downstream of the compressor (1) can be used to
optimize or protect the compressor (1).

5.   The method for controlling evaporators (4) in
refrigeration plants as claimed in claims 1-4,
characterized in that, as mentioned above, control is

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effected, optimally for the particular type of evaporator, near to the left-hand limit curve of the lg p, h diagram for refrigerant (12).

5   6.   The method for controlling evaporators (4) in refrigeration plants as claimed in claims 1-5, characterized in that this type of control causes the evaporator (4) to be flooded and the degree of flooding to be determined, and at the same time causes the

10   refrigerant suction vapor temperature and refrigerant liquid temperature (13/11) to be monitored and controlled.

7.   The method for controlling evaporators (4) in

15   refrigeration plants as claimed in claims 1-6, characterized in that the measured value for limiting the suction vapor temperature (13/14) upstream of the compressor (1) (or the corresponding alternative measured values 14, 15, 16 as described above) over-

20   controls the evaporation control (11, 12) and keeps the suction vapor temperature (14) constant at an optimum and/or maximum value as a function of the compressor.

8.   The method for controlling evaporators (4) in

25   refrigeration plants as claimed in claims 1-7, characterized in that the optimum of the process is always in favor of the evaporator (4) and not the IHE (5) (maximum utilization of the enthalpy in the evaporator (4) between the left-hand and right-hand

30   limit curves of the lg p, h diagram for refrigerant and, depending on the temperature level of the IHE (5) (21), with a superheating component in the evaporator (4)).

35   9.   The method for controlling evaporators (4) in refrigeration plants as claimed in claims 1-8, characterized in that one evaporator (4) can be connected to one IHE (5), or a plurality of evaporators

(4) can be connected to one IHE (5) or a plurality of
evaporators (4) can be connected to a plurality of IHEs
(5), or any type of combinations thereof, to form a
refrigeration system (Fig. 10-18).

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10. The method for controlling evaporators (4) in
refrigeration plants as claimed in claims 1-9,
characterized in that, depending on the combination of
evaporators (4), IHEs (5), injection valves (3) and
10   compressors (1), the injection valve (3) and the system
can be controlled with reduced measured values (9, 10,
11, 12, 13, 14, 15, 16, 22, 23) (in each case 1
measured value (9, 10, 11, 12, 13, 14, 15, 16, 22, 23)
for in each case one expansion valve (3) (11, 12)
15   and/or compressor (1) (13, 14, 15, 16), or in each case
1 measured value (9, 10, 11, 12, 13, 14, 15, 16, 22,
23) for a plurality of expansion valves (3) (11, 12)
and/or compressors (1) (13, 14, 15, 16) and/or the
combination of one and/or more measured values (9, 10,
20   11, 12, 13, 14, 15, 16, 22, 23) (Fig. 10-18)).

Fig. 1

Fig. 2

Fig. 3

P_c, t

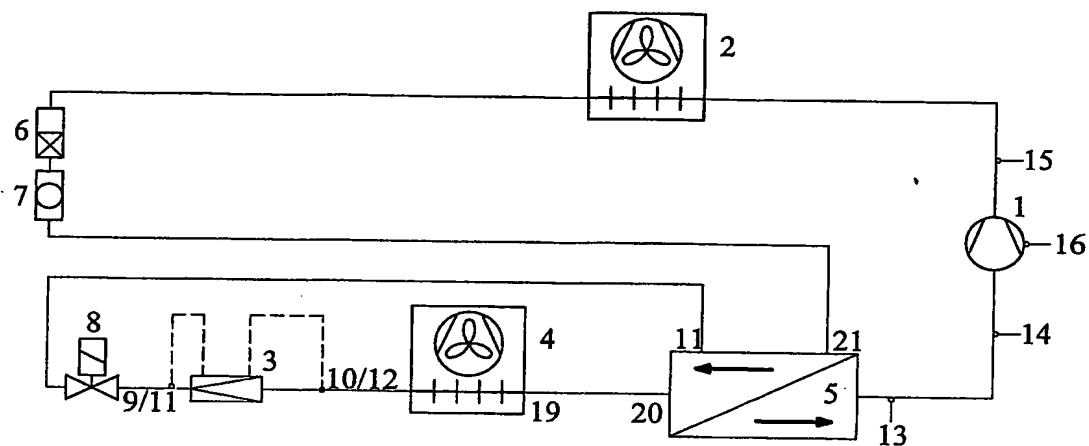P_o, t_o

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12

13

5

Fig. 4

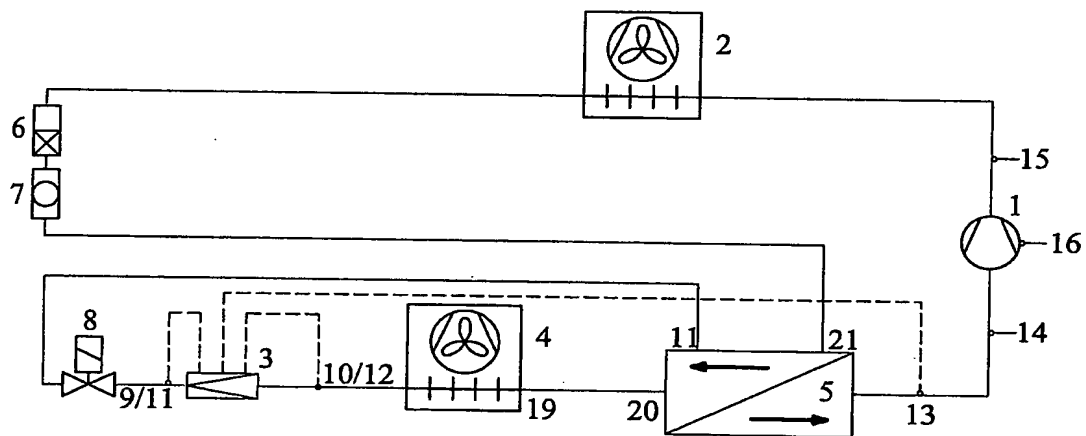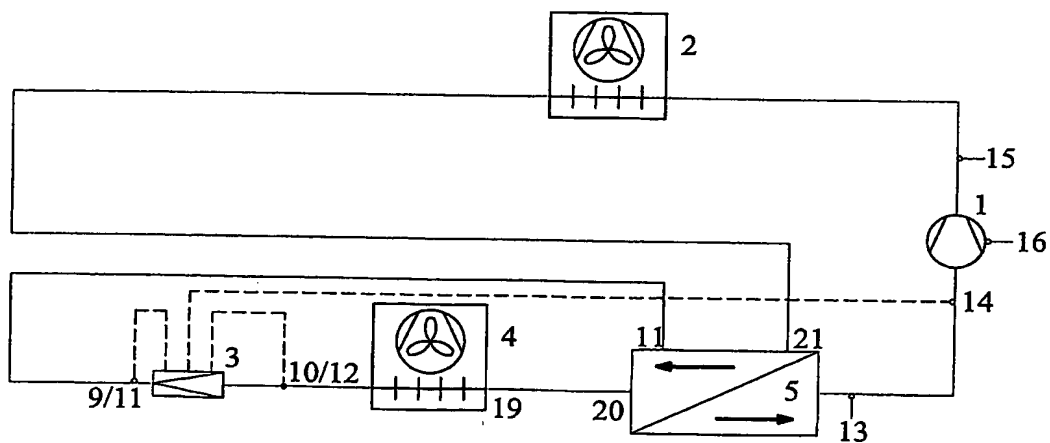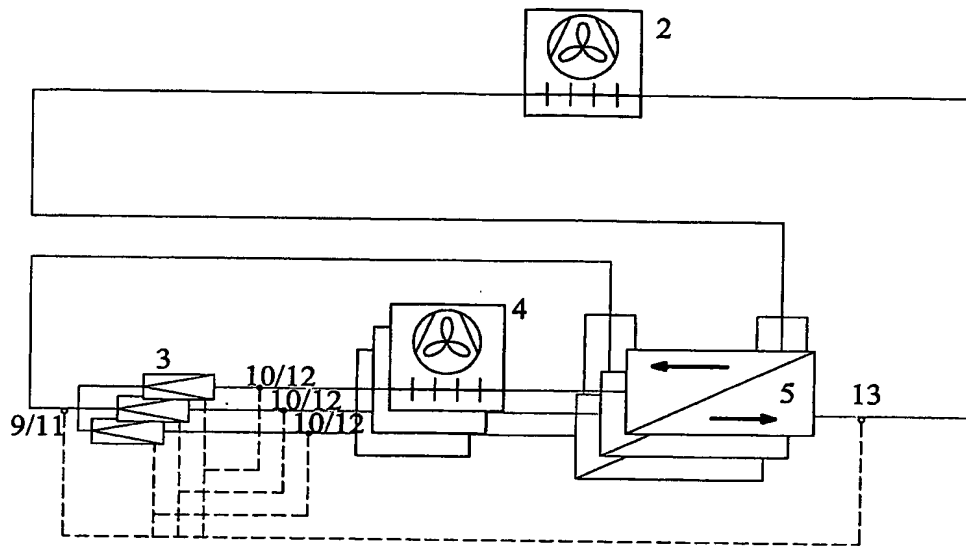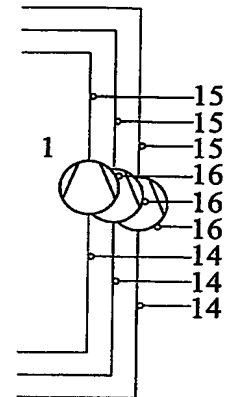Fig. 5

Fig. 6

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Fig. 7.

Fig. 8

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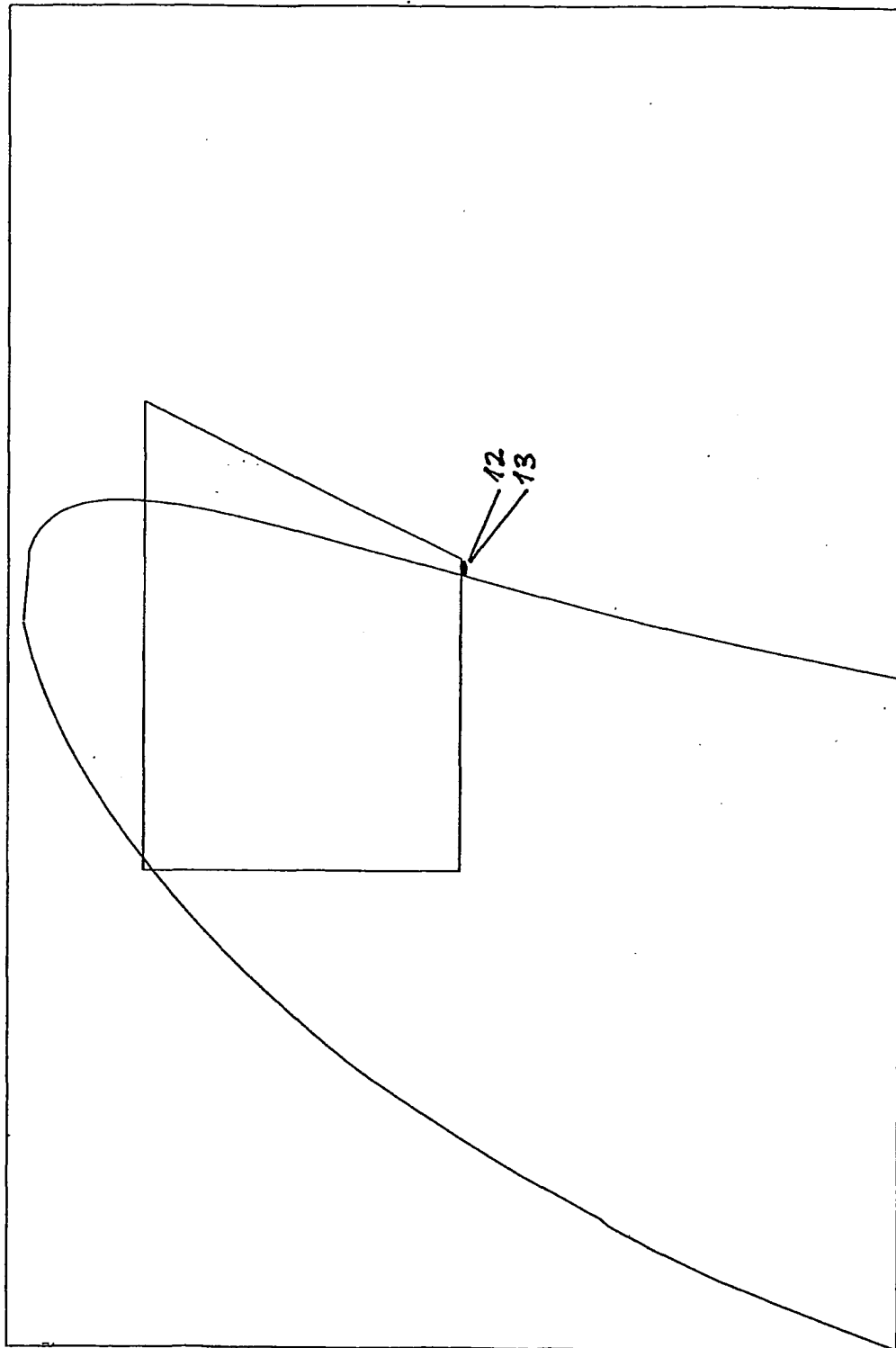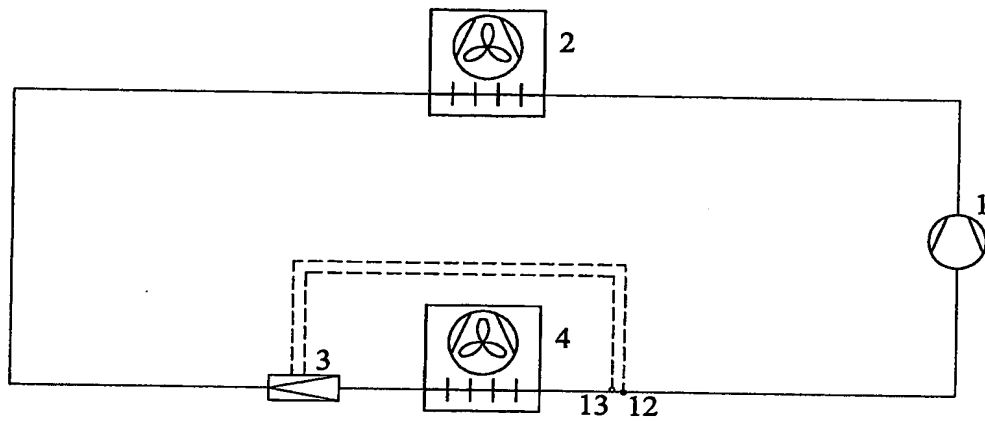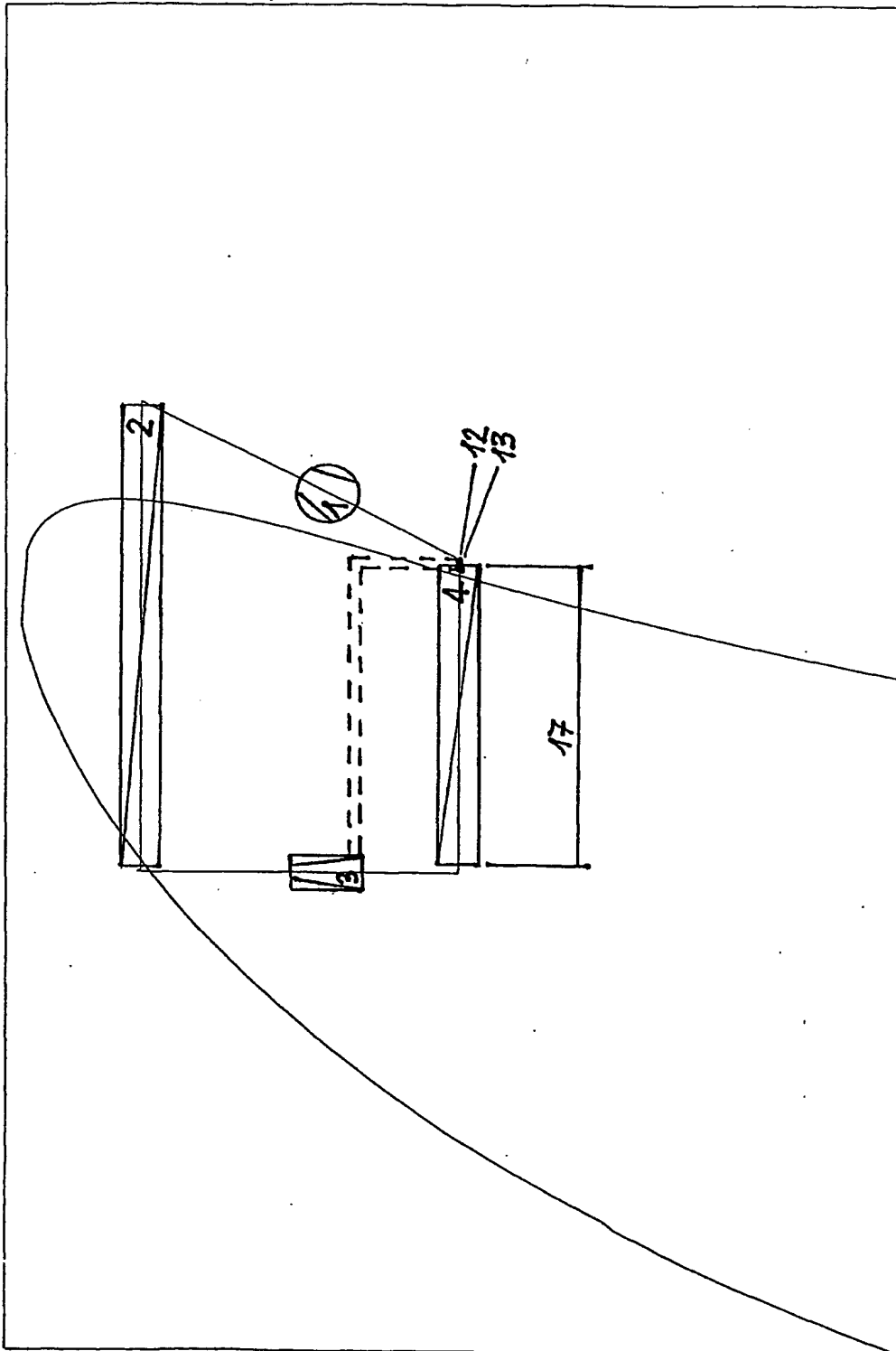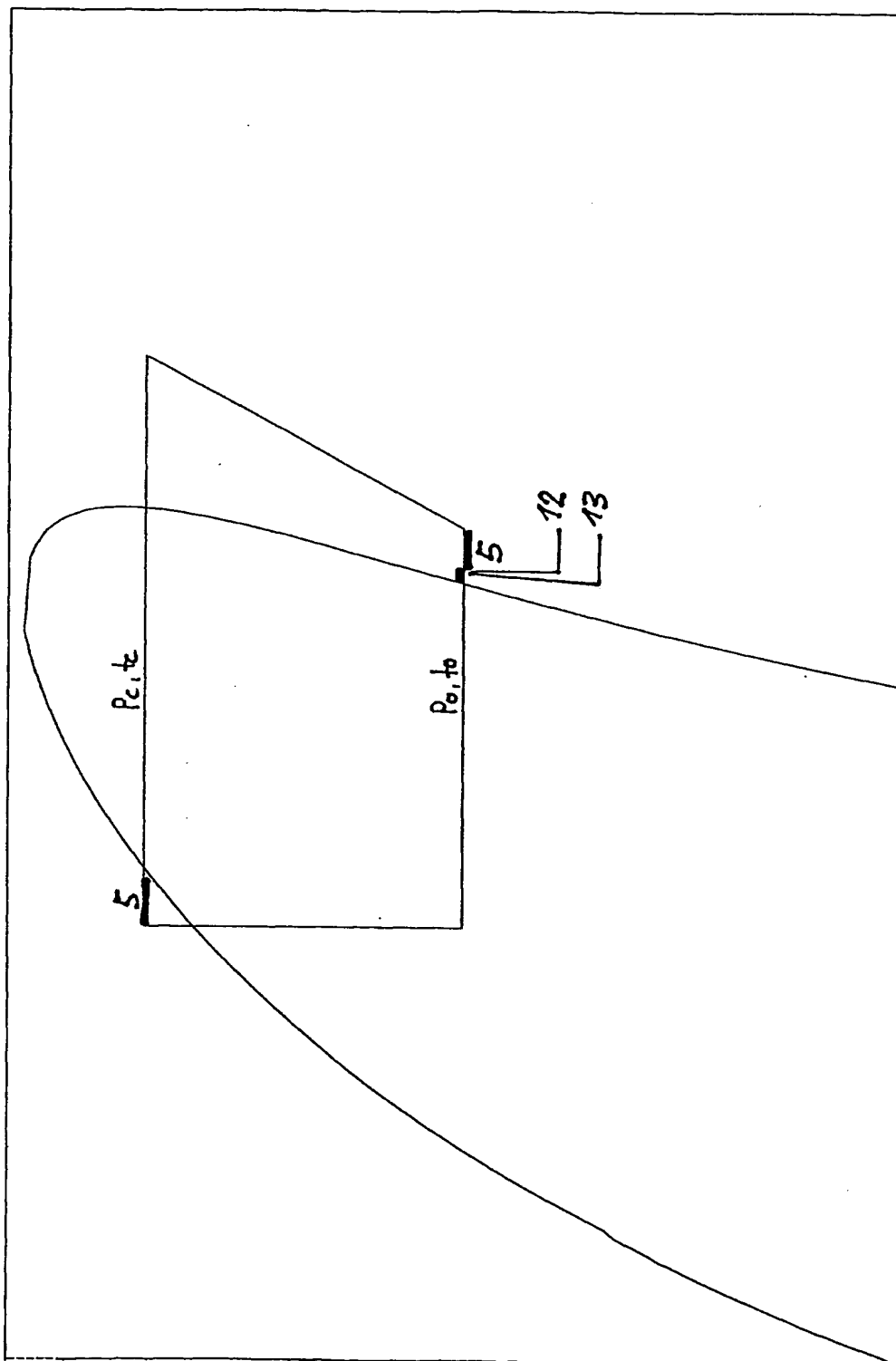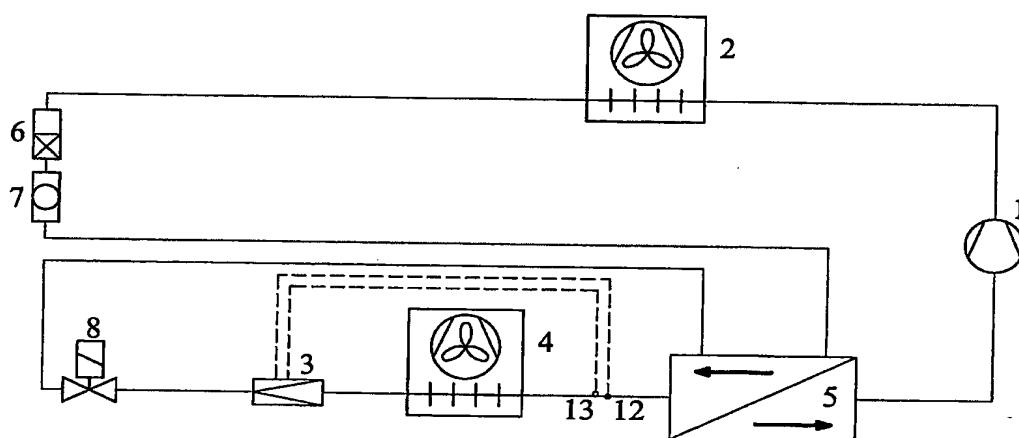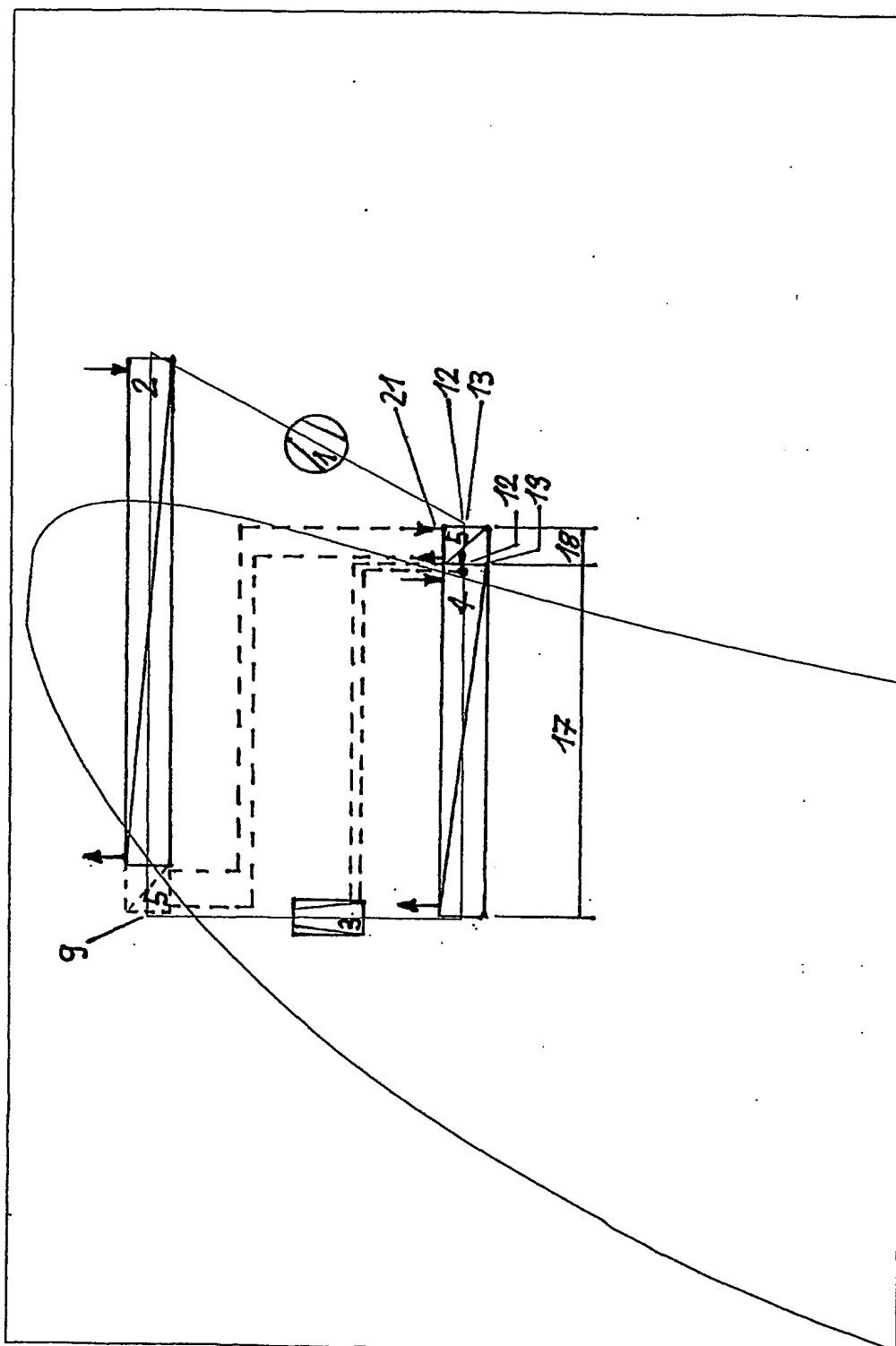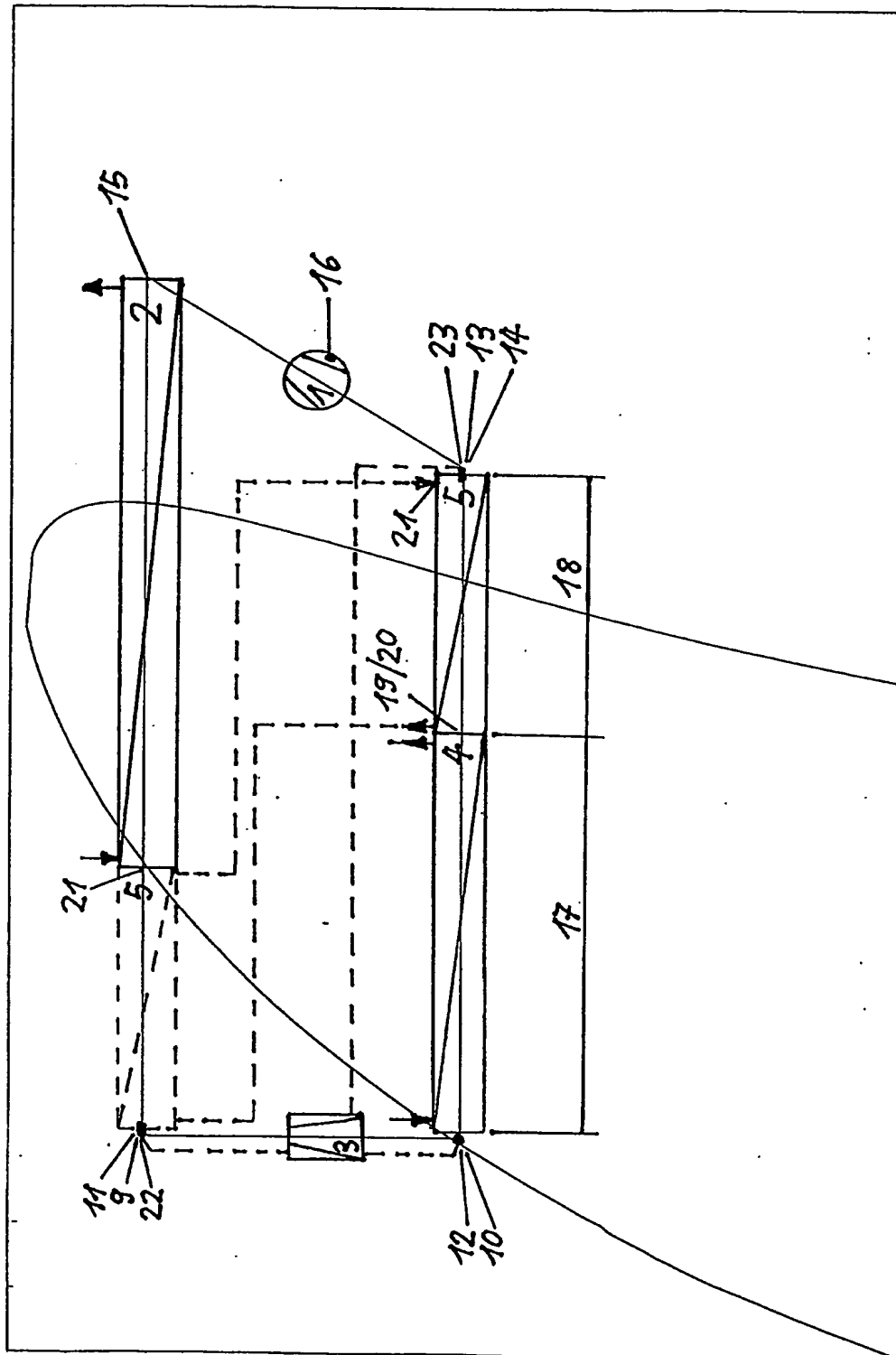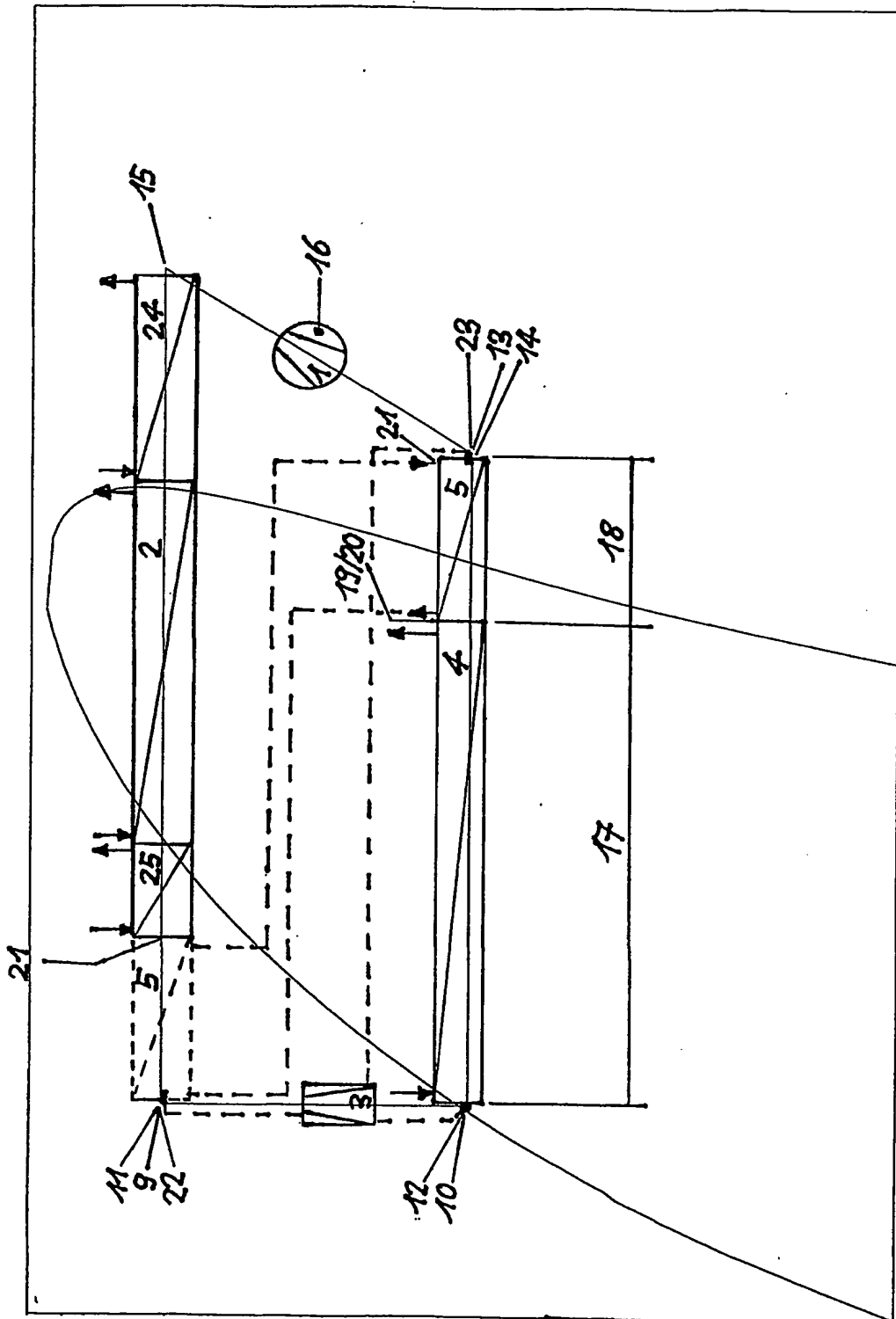
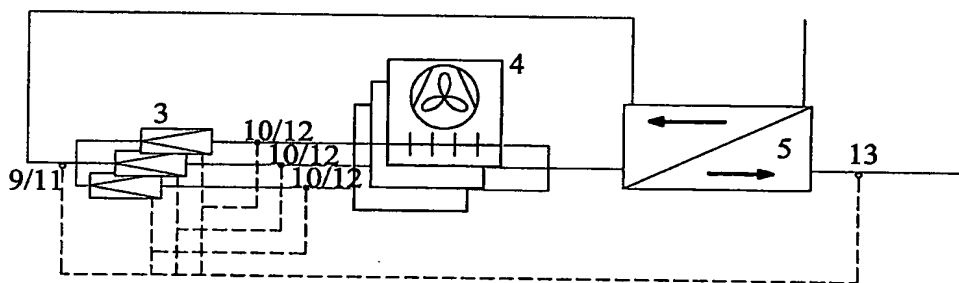Fig. 9

Fig. 10

Fig. 11

Fig. 12

Fig. 13

Fig. 16

Fig. 14
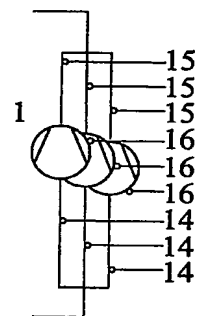
Fig. 17
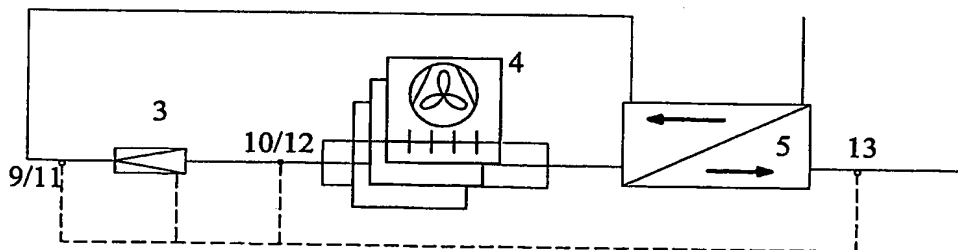
Fig. 15

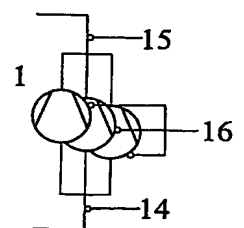Fig. 18

Legend for the points from the drawings:

- 1   compressor
- 2   liquefier
- 3   injection valve, expansion valve
- 4   evaporator
- 5   IHE (internal heat exchanger)
- 6   drier
- 7   inspection window
- 8   solenoid valve
- 9   temperature sensor at the liquid line upstream of the injection valve
- 10   pressure sensor at the evaporator inlet (process)
- 11   refrigerant liquid temperature upstream of the injection valve
- 12   suction vapor pressure, evaporation pressure
- 13   suction vapor temperature at the evaporator or IHE exit
- 14   suction vapor temperature at the compressor inlet
- 15   hot-gas temperature at the compressor outlet
- 16   compressor oil temperature
- 17   evaporation enthalpy (evaporator power)
- 18   IHE enthalpy (IHE power)
- 19   x point of the state of the refrigerant in the wet steam region (kg/kg), end of evaporator
- 20   x point of the state of the refrigerant in the wet steam region (kg/kg), IHE entry
- 21   IHE entry temperature of the liquid refrigerant
- 22   high pressure
- 23   suction pressure upstream of the compressor
- 24   deheater (optional)
- 25   external supercooler (optional)

Fig. 19